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Wen-Foo Yau and William S. Durant

E. I. du Pont de Nemours & Co.
Savannah River Laboratory
Aiken, SC 29808-0001

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RESPONSE OF A GLASS MELTER TO STEAM EXPLOSION*

Wen-Foo Yau and William S. Durant

E. I. du Pont de Nemours & Co., Savannah River Laboratory
Aiken, South Carolina 29808-0001, USA

INTRODUCTION

The U.S. Department of Energy is proposing to build the nation's first Defense Waste Processing Facility (DWPF) at the Savannah River Plant (SRP) for large-scale immobilization of high-level radioactive wastes. The SRP is the primary producer of radionuclides in the country for nuclear weapons. In the past 26 years of SRP operations, 28 million gallons of nuclear wastes have been generated, and waste generation is expected to continue at a rate of greater than 1.5 million gallons per year. This waste is currently stored in large underground tanks at the SRP: the DWPF will provide a timely solution to the problem of nuclear waste disposal. The method of waste immobilization calls for a two-stage process that will solidify the sludge and salt fractions of the waste separately. In the first stage, the sludge fraction of the waste is solidified with a borosilicate-type glass frit in a joule-heated ceramic melter at temperatues of about 1150°C. The molten glass is then poured into stainless-steel canisters for interim storage. The stage 1 project will include interim storage facilities at the SRP; thus making operation of the DWPF independent of the eventual selection and timing for a federal repository.

The glass melter (Figure 1) for the continuous pouring process is of multilayered design using a combination of structural materials to accommodate the severe thermal loading under normal operating conditions. The principal containment structure is a stainless steel shell which is cooled from the outside by a water jacket and insulated from inside by a layer of refractory blocks. Between the refractory and the steel

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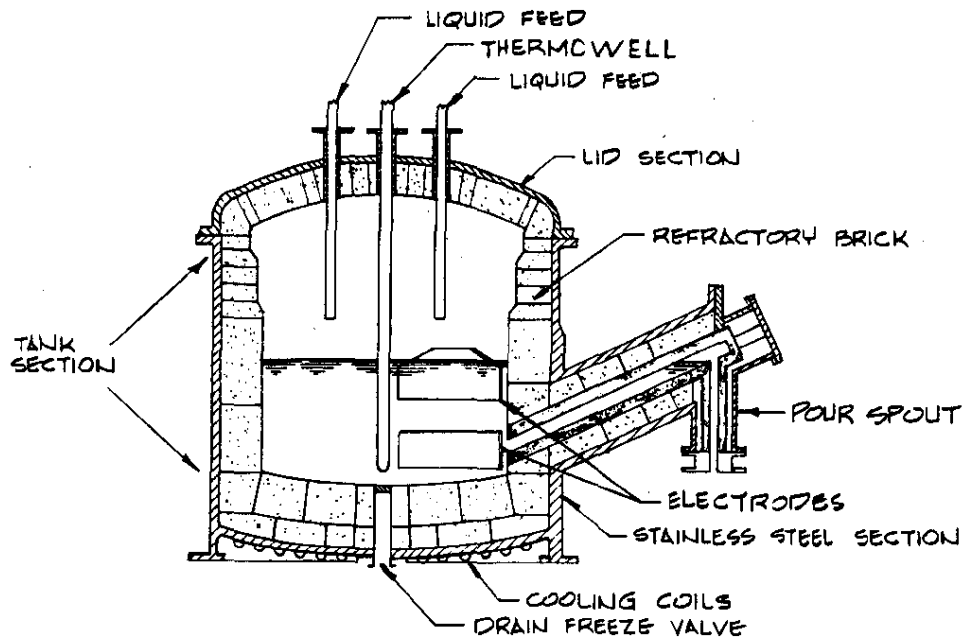

Figure 1. DWPF Glass Melter

shell, there is a thin layer of porous material serving to absorb the outward expansion of the refractory layer at high temperatures. For quality assurance and operational safety of the glass melter, design considerations include the dynamic responses of the melter under abnormal loading conditions, such as that caused by a steam explosion (due to moisture in waste sludge). For assessment of structural integrity of the melter under sudden pressure, this paper provides a conservative estimation of the loading history and analyzes the resistance of the containment structure based on the classical thin shell theory.

ENERGY RELEASE FROM A STEAM EXPLOSION

The efficiencies of steam explosions, based on percentage of the total thermal energy, range from a theoretical maximum of about 30% downward to a fraction of one percent. In general, the larger the amounts of material involved, the lower the efficiency. Based on intermediate scale experiments of Buxton and Benedict (1979), efficiencies in 1.34% downward were reported. Particular tests simulating the thermal and geometrical conditions of the DWPF melter conducted by Hutcherson (1983) indicate that the energy release from a contained steam explosion in a glass melter cannot be greater than one half of one percent of the thermal energy.

Thermal energy is comprised of two elements, the sensible heat and the heat of fusion. The calculation is based on the physical properties and operating conditions shown in Table 1. Sensible heat is computed as the product of the mass of glass, its average specific heat and the difference between operating and transition temperatures. The heat of fusion of the waste glass is estimated to be 3400 cal/mole like that for SiO_2 quartz. It can be shown that the sum of these terms is $2.7(10)^6$ pcu or 5125 MW-sec.

Table 1

Physical Properties and Operation Conditions for Molten DWPF Glass

Operating temperature:	1150°C
Mass of glass:	10,241 pounds
Transition temperature:	459°C
Average specific heat:	0.3 pcu/lb - °C
Heat of fusion:	3400 cal/mole
Distance from center of melter to Mullfrax wall:	3 ft
Distance from center of melter to Mullfrax lid:	3.5 ft

Thermal energy is expressed in terms of TNT equivalence to enable scaling properties to be applied in the determination of pressures at the melter wall according to Proctor (1966). The energy released by one pound of TNT is approximately 2 MW-sec; thus, 5125 MW-sec is equivalent to 2652 pounds of TNT. Base on a 0.5% efficiency the explosion would involve 12.8 pounds of TNT equivalence. It is assumed that the explosion is initiated from the melt surface at a point in the center of the melter. From Figure 2 for the attenuation of TNT explosions in air, the pressure at 3 feet to the wall would be 410 psig and at 3-1/2 feet to the lid would be 300 psig.

Pressure rise time, that is the time for the pressure to increase from 10% to 90% of maximum value, is estimated to be 100 microseconds based on approximate rise time of 100, 100 and 150 microseconds for steam explosions reported by Buxton, Nelson, and Benedict (1979), TRW Space Technology Laboratory (1964), and Argonne National Laboratory (1964). A pulse duration of 1 millisecond is estimated from the data of Buxton, Nelson, and Benedict. These parameters are used in estimating pressure impulses.

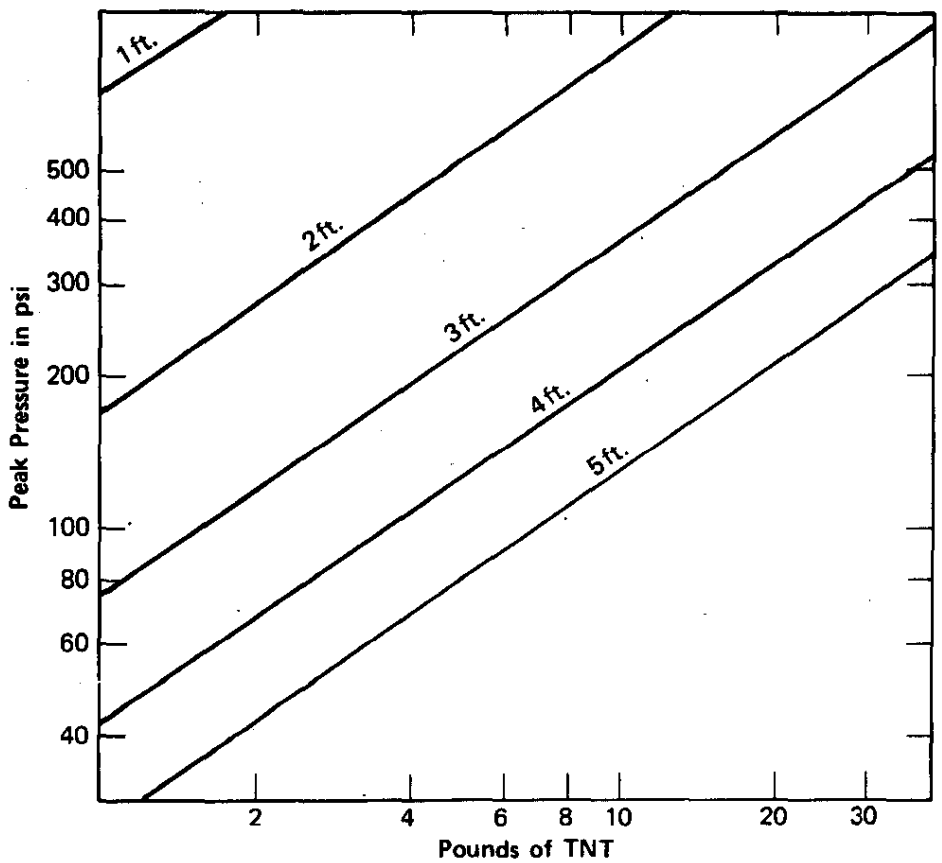

Figure 2. Attenuation of TNT Explosion Pressure in Air

DYNAMIC RESPONSES OF THE MELTER STRUCTURE

The steel containment structure of the melter consists of two spherical heads attached to a cylindrical wall, and all three shells have large ratios of radii of curvature to wall thicknesses such that the thin shell theory is applicable for analysis. The dynamic pressure resulting from a possible steam explosion is assumed to exert on the steel shell directly without considering dissipation due to the melter contents, such as the refractory blocks, electrodes, and molten glass. Furthermore, the time history of pressurization is assumed to be the same for all shell components except that the maximum pressure on the cylindrical wall, P_{1m} = 410 psi, is greater than those of the spherical shells, P_{2m} = 300 psi. (The postulated site of explosion is the geometrical center of the melter; the cylindrical wall is closer to the center of explosion than the top and bottom heads, so it experiences a higher peak pressure.) A triangular impulse function f(t) is used to describe the time history:

$$f(t) = \begin{cases} t/t_m & t_m > t > 0 \\ (t_1 - t)/(t_1 - t_m) & t_1 > t > t_m \\ 0 & t > t_1 \end{cases} \qquad (1)$$

where the time t is measured in seconds, $t_1 = 0.001$ sec is the duration of the impulse, and $t_m = 0.1t_1$, is the time when the peak impulse occurs.

In the analysis the dynamic response of each shell component is first determined in terms of its radial displacement and membrane stress, then bending stresses (or discontinuity stresses) are calculated at the boundary where two adjacent shells join by use of the conditions of geometrical compatibility. Of the two spherical heads of the melter, the bottom one is designed with a larger radius of curvature than the top shell, so the membrane stress in the bottom head is always greater. For purposes of estimation of design safety, it is sufficient to analyze the stresses in the region where the cylindrical wall joins the bottom shell. Using subscripts 1 and 2 to denote quantities pertaining to the cylindrical wall and the bottom shell respectively, it can be shown the dynamic responses in terms of the membrane stresses σ_1 and σ_2 are governed by:

$$\frac{d^2\sigma_1}{dt^2} + \frac{E}{\rho r_1^2}\sigma_1 = \frac{E}{\rho r_1 h_1} P_{1m} f(t) \qquad (2)$$

$$(1-\nu)\frac{d^2\sigma_2}{dt^2} + \frac{2E}{\rho r_2^2}\sigma_2 = \frac{E}{\rho r_2 h_2} P_{2m} f(t) \qquad (3)$$

where $E = 30\ (10)^6$ psi, $\rho = 7324\ (10)^{-4}$ lb-sec^2/in^4 and $\nu = 0.3$ are the modulus of elasticity, mass density and Poisson's ratio of stainless steel respectively, the radii of curvature are $r_1 = 48.75$ inches and $r_2 = 197$ inches, and the shell thicknesses are $h_1 = 1.500$ inches and $h_2 = 1.125$ inches. The solutions to Equations 2 and 3 can be written in an abbreviated integral form by:

$$\sigma_i = s_i\ \omega_i \int_0^t f(\tau) \sin \omega_i (t - \tau)\, d\tau \qquad (4)$$

where i = 1 or 2, $s_1 = P_{1m} r_1/h_1 = 13.3$ ksi, $s_2 = P_{2m} r_2/(2h_2) = 26.3$ ksi, $\omega_1 = \left(E/(\rho r_1^2)\right)^{1/2} = 4152$ rad/sec. and $\omega_2 = \left\{2E/\left((1-\nu)\rho r_2^2\right)\right\}^{1/2} = 1737$ rad/sec. Those membrane stresses together with the loading impulses are plotted in Figure 3. Since structural damping, which is not included in the formulation, tends to reduce responses in cyclical motion, only the first cycle stress levels are shown in the figure.

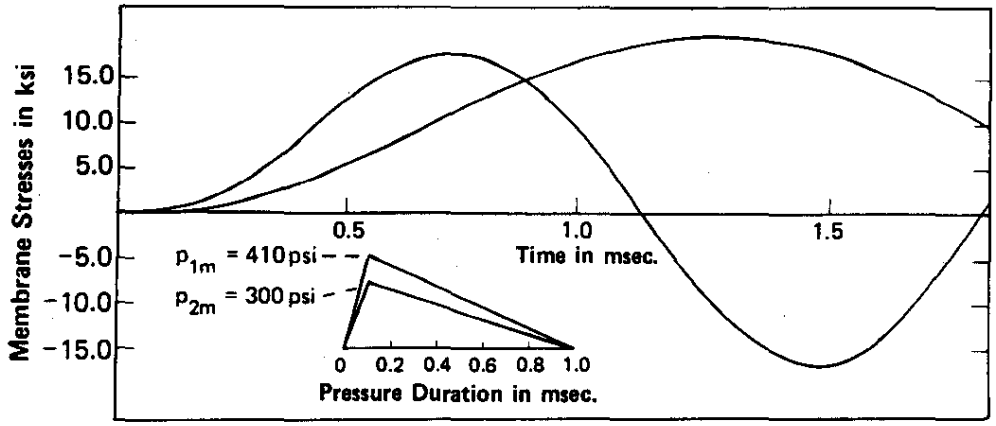

Figure 3. Loading and Response Histories

The results of individual shell responses indicate certain mismatches of displacement at the boundary where the cylindrical shell joins the bottom head. As shown schematically in Figure 4a, the horizontal component of the radial displacement for the spherical shell u_{2x} is not the same as the radial displacement for the cylindrical shell u_1. In order to match these displacements at the boundary, a bending moment M and a shearing force Q must be applied to both shell edges in opposite directions, as shown in Figure 4b. It can be shown that for a compatible edge displacement and edge rotation, M and Q must satisfy

$$Q + 2\lambda_1 M - \frac{\lambda_2^2}{\lambda_1^2} \cdot \frac{h_1}{h_2} \cdot \frac{1}{r_1 r_2} \left(Q - \frac{2\lambda_2}{r_1} M\right) = 0$$

$$Q + \lambda_1 M + \frac{\lambda_2}{\lambda_2} \cdot \frac{h_1}{h_2} \cdot \frac{1}{r_2} \left(Q - \frac{\lambda_2}{r_1} M\right) = \frac{h_1}{2\lambda_1 r_1} (\sigma_2 - \sigma_1) \qquad (5)$$

where $\lambda_1 = \left(3\,(1-\nu^2)/(r_1 h_1)^2\right)^{1/4} = 0.1503/\text{inch}$, and

$$\lambda_2 = \left(3(1-\nu^2) r_2^2/h_2^2\right)^{1/4} = 17.01$$

The solution to Equations 5 is dependent upon the difference of membrane stresses $(\sigma_2 - \sigma_1)$. From Figure 3, the maximum difference occurs at t = 0.00152 sec, i.e.,

$$/\sigma_2 - \sigma_1/\text{max} = 17.6 - (-16.8) = 34.4 \text{ ksi}$$

The corresponding edge moment and shear at that time are

M = 1043 inch-lb/inch
Q = 2067 lb/inch

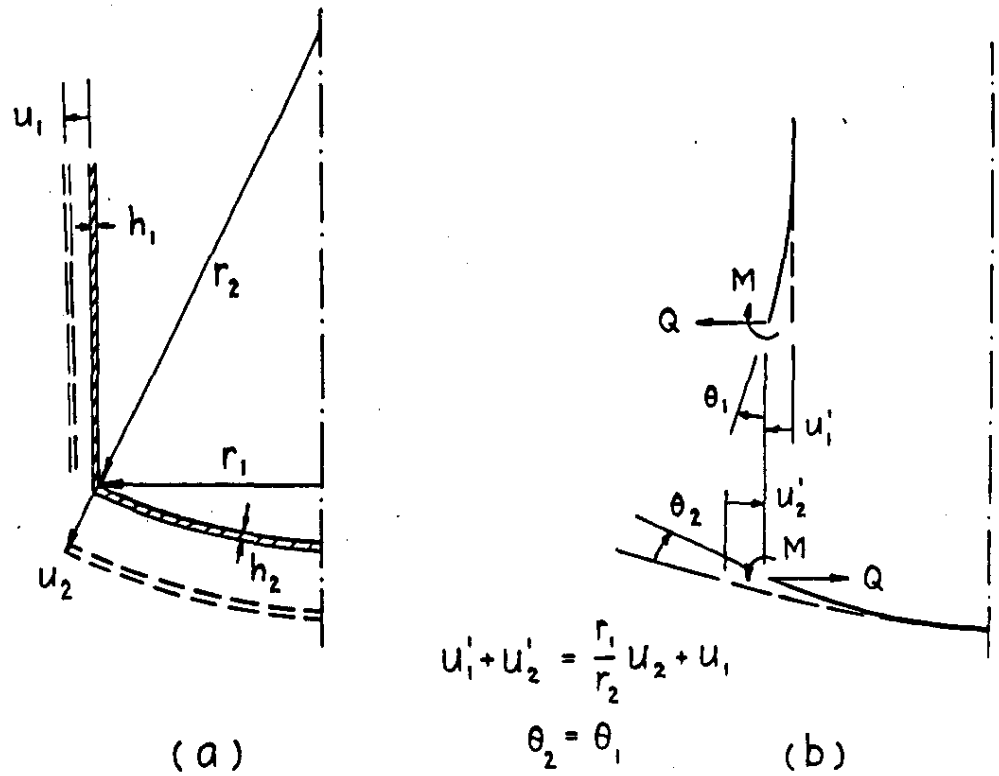

Figure 4. Boundary Conditions at Shell Joint

and the bending stress in the cylindrical and spherical shells are, respectively,

$$\sigma_{1b} = 6M/h_1^2 = 2.78 \text{ ksi}, \qquad \sigma_{2b} = 6M/h_2^2 = 4.94 \text{ ksi}$$

The time when the bending stress σ_{2b} reaches its maximum differs slightly from the time when the membrane stress σ_2 is the maximum; they are assumed to coincide for conservative estimation, then the dynamic stress due to the postulated steam explosion is 24.2 ksi. Stainless steel 304L has a minimum yield stress of 25 ksi; its yield strength under impact loading conditions is generally higher, so the shell structure of the melter is considered as elastically resistant to the sudden pressure of a postulated steam explosions.

SUMMARY

As part of the safety assessment in the design of the glass melter for large-scale immobilization of high-level radioactive wastes, structural considerations of the containment shell include its dynamic responses to abnormal loading conditions such as that caused by a steam explosion. The postulated steam explosion, conservatively given an energy content

equivalent to 13 pounds of TNT, is capable of exerting sudden pressures greater than 300 psi but less than 410 psi on the melter wall. By use of thin-shell theory, the equations of motion satisfying the discontinuity conditions at junctions of shells with different curvatures are solved analytically. Results of stress analysis ensure elastic responses of the containment structure of the melter.

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